(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
22 April 2004 (22.04.2004)

PCT

(10) International Publication Number
WO 2004/033906 A1

(51) International Patent Classification⁷: F04B 1/22, 49/02, F01B 3/02, 31/22

(21) International Application Number: PCT/AU2003/001337

(22) International Filing Date: 10 October 2003 (10.10.2003)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
2002951972 10 October 2002 (10.10.2002) AU

(71) Applicant (*for all designated States except US*): PERMO-DRIVE RESEARCH AND DEVELOPMENT PTY LTD [AU/AU]; 203 Southern Cross Drive, Ballina, NSW 2478 (AU).

(72) Inventor; and
(75) Inventor/Applicant (*for US only*): RUSH, Allan [AU/AU]; 65 Booerie Creek Road, Lismore, NSW 2480 (AU).

(74) Agent: BALDWIN SHELSTON WATERS; 60 Margaret Street, Sydney, NSW 2000 (AU).

(81) Designated States (*national*): AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NI, NO, NZ, OM, PG, PH, PL, PT, RO, RU, SC, SD, SE, SG, SK, SL, SY, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, YU, ZA, ZM, ZW.

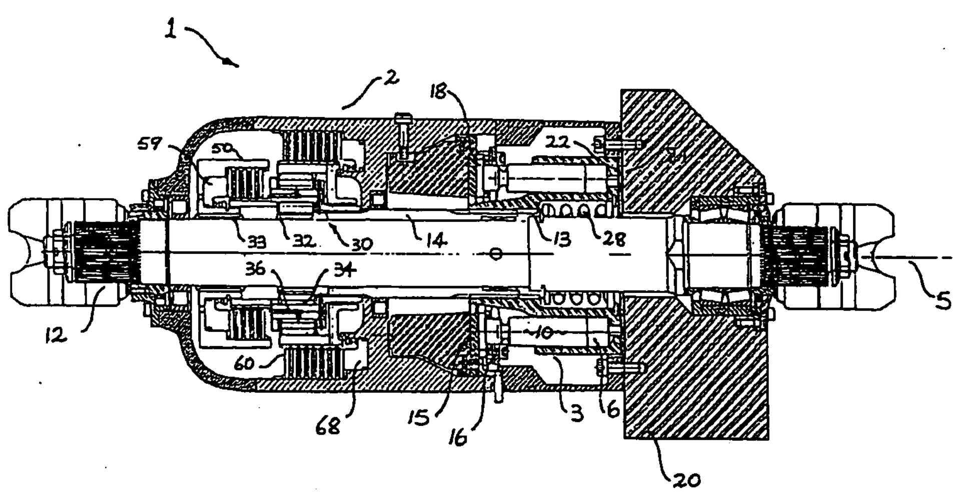
(84) Designated States (*regional*): ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IT, LU, MC, NL, PT, RO, SE, SI, SK, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Declaration under Rule 4.17:
— *of inventorship (Rule 4.17(iv)) for US only*

Published:
— *with international search report*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) Title: HYDRAULIC PUMP/MOTOR WITH EPICYCLIC GEAR CONTROL



(57) Abstract: A positive displacement hydraulic pump/motor (1) assembly (1) includes a rotary cylinder block (3) a circular array of cylinders (6) around a central axis (5) with corresponding pistons (10) disposed within. Drive shaft (12) extends through a bore (13) and rotates the cylinder block (3) while a stationary valve plate (20) having a valve face (21) mating with a complementary mating surface (22) formed on the cylinder block (3). The valve plate (20) provide for inlet and outlet ports for the fluid communication with a source of hydraulic fluid and a hydraulic load. A coupling mechanism incorporating an epicyclic gear train (30) is operable in an engaged mode to connect the drive shaft (12) to the cylinder block (3).

WO 2004/033906 A1

HYDRAULIC PUMP/MOTOR WITH EPICYCLIC GEAR CONTROL

FIELD OF INVENTION

The present invention relates generally to hydraulic motors and pumps.

The invention has been developed primarily for use with a positive displacement pump/motor assembly which forms part of a regenerative drive system ("RDS"), and will be described predominantly hereinafter in that context. It should be appreciated, however, that the invention is not limited to this particular field of use, being readily adaptable to any axial hydraulic motor or pump for use in virtually any application.

BACKGROUND OF THE INVENTION

In one particularly beneficial context, the invention has been developed more specifically as an improvement to the RDS described by the present applicant in an earlier patent application filed via the Patent Cooperation Treaty (PCT) as international application No PCT/AU99/00740, the full contents of which are hereby incorporated by reference.

As previously described in that earlier patent application, the RDS is based upon a positive displacement hydraulic pump/motor arrangement incorporating a cylinder block which houses a cylindrical array of axially reciprocating pistons. In the preferred embodiment, the cylinder block and valve face are coaxially disposed around the primary drive shaft. When used in conjunction with a suitable accumulator, the resultant regenerative drive system provides a practical and commercially viable system for harnessing the previously wasted braking energy of a vehicle (or other system), storing this energy, and subsequently releasing it back into the drive train as required under conditions of acceleration, peak load, or gear change transitions.

When applied to a vehicle, which is only one of numerous potential applications, this RDS arrangement significantly improves the overall efficiency of the engine and power transmission systems. The RDS system also conveniently acts as an efficient auxiliary braking mechanism in the energy accumulation mode. The system as described, however, is subject to several limitations.

One significant limitation in the pump/motor assembly as previously described, relates to the inherent drag associated with the seals, bearings, valve faces and other elements that are in direct sliding contact with each other, as the cylinder block and other components in the rotational group rotate with respect to the valve plate, housing, oil bath and other stationary components of the system. In the particular RDS unit described, these frictional forces are reduced by virtue of the fact that most of the components in direct contact with one another "float" on a film of lubricating oil, which is ideally pressurised. Nevertheless, a residual drag factor remains, which consumes energy, generates heat, accelerates wear and generally compromises the potential efficiency of the system.

From this perspective, the present invention is concerned, among other things, with the drag normally attributable to the direct communication between the rotational group including the cylinder block, and the valve plate. In the pump/motor assembly and RDS unit as described in the earlier patent application, a "hold-down" spring is disposed resiliently to bias the cylinder block axially into face-to-face sliding engagement with the valve plate. Because of the relatively high pressures generated within the pump/motor assembly, the hold-down spring is required to exert a relatively high degree of axial force on the cylinder block in order to prevent excessive fluid leakage between the block and the valve plate, particularly under conditions of maximum pressure or power transmission.

While effective in preventing excessive leakage, the relatively high axial force between the cylinder block and the valve plate causes significant frictional drag between these components as they slide rotationally relative to one another, separated only by a thin film of pressurised oil. From an overall system perspective, the associated inefficiency is proportionally more significant in situations where the pump/motor assembly is operating in a neutral or "free-wheeling" mode. This is because in that condition, the unit is neither pumping nor driving and is therefore doing no real work against which its own inherent inefficiency can be offset, and must nevertheless continue to rotate, often at high speed, as a result of its direct connection with the primary power source.

While applying to some extent in almost any application of positive displacement hydraulic motors and pumps, this limitation is particularly significant in

the context of an RDS unit fitted to a vehicle, in the manner previously described. That is because in many situations, the RDS may "free-wheel" for cumulative periods of many hours. During this time, the unit may generate neither motor nor pump pressure, but must nevertheless continue to rotate at the same speed as the vehicle drive line to which it is integrally connected. This would predominantly occur, for example, during long runs on open highways with minimal changes in traffic conditions or road topography. In such situations, the RDS may do no effective work for prolonged periods, and yet introduce an inherent drag factor into the drive line, which ultimately compromises the efficiency of the power train of the vehicle.

This basic problem is compounded in such conditions, because the drag factor itself is exacerbated when the pump/motor unit is operating in the neutral or free-wheeling mode. This is because the unit is not automatically producing hydraulic pumping pressure sufficient to sustain the film of hydrostatic pressurised oil, which is required to minimise the effect of frictional drag between the components in direct face to face rotational contact with one another. In such circumstances, an internal or external charge pump may be utilised to supply a 'pilot' oil pressure to maintain lubrication between these components. However, this merely shifts the problem so that the energy losses are then associated with running the supplementary pump. The same situation applies if a pilot pressure for lubrication is created by a small non-zero swash plate angle, or drawn from the accumulators.

Drag is also exacerbated in such applications due to the fact that the cylinder barrel rotates in an oil bath within the housing. While the oil bath provides necessary cooling and lubrication, it is also itself a source of hydrodynamic drag on the integral drive line, which is inefficient when the pump is doing no work.

A further limitation of the earlier RDS relates to the fact that because the drive shaft for the hydraulic/motor unit is effectively integral with the drive line of the vehicle (or other power transmission application), the rotational speed of the cylinder block is effectively limited to the rotational speed of the drive shaft. A number of inherent problems and difficulties flow from this. Firstly, in semi-trailers and other large freight haulage vehicles, the drive lines typically rotate at comparatively low speeds such that small imbalances in the rotational components of the pump/motor unit do not cause significant problems. However, in other applications including motor cars, where the

drive lines are designed for significantly higher rotational speeds, even small out of balance masses in the rotational group can cause significant problems in terms of the vibration, noise, inertial stresses, and potential fatigue failure of critical components. In such applications, it may be desirable to limit the maximum speed of the pump/motor unit to less than the maximum speed of the associated drive line.

One solution to this problem might be to separate the drive shaft of the pump from the drive line by means of a transmission system with an appropriate speed reduction capability. However, this would inevitably increase the size, weight and cost, reduce the efficiency and reliability, and compromise the retrofitability and overall commercial viability of the system.

Attempts have also been made in other contexts to achieve such a sustained speed differential using clutches, regulated to allow a predetermined rate of constant slippage. However, known clutches are only designed to facilitate relatively rapid transitions between different states of speed or gearing. They are generally not suitable to maintain constant speed differentials through partial slippage in dynamic equilibrium conditions. In such situations, they are notoriously difficult to control accurately, and quickly burn out. They also generate significant heat due to internal friction when only partially engaged, and are therefore energy inefficient in this mode.

In other applications, it is in fact desirable for the pump/motor unit to rotate at a speed significantly greater than that of the drive shaft and the associated drive line. For example, in vehicular applications involving frequent stops and starts, such as a garbage collection truck on a suburban run. In this environment, the average vehicle speed is relatively low. Additionally, there may only be a relatively short period of time in which to charge the accumulators as the truck slows down on approach to one collection point, before full acceleration is again required on take-off toward the next collection point only a small distance away. Under these conditions, a pump/motor unit limited to the rotational speed of drive line may not be able to pump sufficient fluid to fully charge the accumulators between stops, even at maximum swash plate angle. The same limitation applies by extension to virtually any situation where rapid charge-up of the system may be required.

There may also be situations in which it would be desirable to rotate the pump/motor unit in the opposite direction to that of the integral drive line, which the

previously described system, for all of its advantages, again does not allow. One reason for this might be to minimise torque reaction in the drive line upon rapid acceleration or braking, by rotating one part of the system in one direction and another part of comparable mass in the opposite direction. Another reason might be to enable full functionality of the RDS unit with the vehicle reversing, without requiring the hydraulic pump/motor unit to rotate in the reverse direction.

The foregoing discussion of the prior art is intended solely to place the invention in an appropriate context, and allow a proper appreciation of its technical significance. Any statements made in this specification about prior art information should not be construed as admissions that such information is widely known, or forms part of common general knowledge in the relevant field.

It is an object of the present invention to overcome or substantially ameliorate one or more of the deficiencies of the prior art, or at least to provide a useful alternative.

DISCLOSURE OF THE INVENTION

Accordingly, the invention provides a positive displacement hydraulic pump/motor assembly including:-

a rotary cylinder block having a central axis and incorporating a generally circular array of cylinders disposed around the axis;

a corresponding plurality of pistons reciprocably disposed within the respective cylinders;

drive means including a drive shaft extending through a bore formed in the cylinder block to effect rotation of the cylinder block about the central axis;

a drive plate disposed at one end of the cylinder block to effect sequentially staggered reciprocation of the pistons in response to rotation of the cylinder block;

a stationary valve plate disposed at an opposite end of the cylinder block, the valve plate having a valve face adapted for sliding rotational engagement with a complementary mating face formed on the cylinder block;

the valve plate further including at least one inlet port adapted for fluid communication with a source of hydraulic fluid and at least one outlet port adapted for fluid communication with an hydraulic load;

the ports being disposed such that in use, hydraulic fluid is progressively drawn into the cylinders in sequence as the respective pistons are displaced away from the valve plate and subsequently expelled from the cylinders as the pistons are progressively displaced toward the valve plate;

the pump/motor assembly further including a coupling mechanism incorporating an epicyclic gear train operable in at least one engaged mode to connect the drive shaft to the cylinder block.

It will be appreciated that the same assembly may be used in one mode as a motor, and in another mode as a pump. It should therefore be understood that throughout the specification, these terms may be used in conjunction, or interchangeably. In each case, however, unless the context clearly dictates otherwise, any reference to configuration of the invention as a pump should be understood to include configurations as a motor, and vice versa. It should also be understood that the inlet and outlet ports may alternate in function according to the mode of operation of the unit.

It should further be understood that the terms "sealing", "sealing engagement" and the like are intended to convey the sense of prevention of excessive leakage past or through the relevant components. It will be appreciated, however, that in a system of this nature, a minimal level of leakage flow may persist, and indeed maybe desirable for lubrication purposes, notwithstanding the fact that effective sealing, in the intended sense, has been achieved.

Preferably, the cylinders and pistons are disposed in generally parallel relationship around the central axis, in an axial piston configuration.

In a particularly preferred embodiment, the positive displacement pump/motor is a swash plate type unit. In this embodiment, the drive plate takes the form of a stationary swash plate, which is inclined with respect to the central rotational axis of the cylinder block. Preferably also, the ends of the pistons remote from the valve plate include "followers" adapted to slide over the swash plate as the cylinder block rotates. A hold-down plate is preferably disposed to capture the floating ends of the pistons and retain the followers in sliding contact with the swash plate. In alternative embodiment, however, springs or other suitable means may be used to retain the followers in contact with the swash plate.

Preferably, the angle of inclination of the swash plate is selectively adjustable, to provide variable flow rate characteristics. In particular, the swash plate is preferably adapted to be selectively inclined in a positive or a negative sense, thereby enabling the assembly alternately to operate as a motor or a pump. Most preferably, the variable swash plate can also be oriented in an intermediate or neutral position, effectively normal to the central axis, such that rotation of the cylinder block causes no movement of the pistons, hence induces no net flow into or out of the cylinders through the ports, and therefore causes no load on the system aside from a residual level of inherent frictional drag.

In other embodiments, it will be appreciated that the invention may also be applied to a bent axis type hydraulic pump. In that case, connecting rods for the pistons are pivotably attached to a thrust plate adapted to rotate with the cylinder block. The invention may also be adaptable to other configurations of motors and pumps.

Preferably, the coupling mechanism is selectively operable in at least two distinct modes, at least one of which is an engaged mode whereby drive is transmitted from the drive shaft to the cylinder block. Preferably also, the coupling mechanism is selectively operable in a disengaged mode, to allow the drive shaft to rotate substantially independently of the cylinder block. In this way, the coupling mechanism preferably also constitutes a decoupling mechanism.

Preferably, the epicyclic gear train includes an input shaft connected to a sun gear, a plurality of planetary gears disposed in meshing engagement around the sun gear, a planetary gear carrier, a peripheral ring gear in meshing engagement with the planetary gears, and an output shaft. It will be appreciated that the nature of the input and output shafts will alternate, depending upon whether the hydraulic unit is operating as a motor or a pump. Preferably, however, one constitutes the primary drive shaft extending coaxially through the pump/motor unit, while the other constitutes a drive sleeve disposed coaxially around the primary shaft.

In the preferred embodiment, the coupling mechanism also includes a clutch and/or a brake assembly, selectively and independently actuable to enable the epicyclic gear train to transmit rotary drive between the drive shaft and the cylinder block in the different modes.

In one particularly preferred technical application, the invention as defined is adapted for incorporation into an energy management system including:-

energy accumulation means operable selectively as the hydraulic load to store and release energy through controlled receipt and release of hydraulic fluid;

a positive displacement hydraulic pump/motor assembly as defined above, in fluid communication with the energy accumulation means;

a low pressure hydraulic reservoir in fluid communication with the pump/motor assembly;

and connection means for connecting the pump/motor assembly to a drive line;

the system being arranged such that in a braking mode the pump/motor assembly retards the drive line by pumping hydraulic fluid into the accumulation means, in a driving mode the pump/motor assembly supplies supplementary power to the drive line using pressurised hydraulic fluid from the accumulation means, and in a neutral mode the pump/motor assembly is effectively inoperative and exerts no substantial driving or retarding influence on the drive line.

In one preferred embodiment, the reservoir includes a low-pressure accumulator or constant pressure chamber adapted to supply a positive pressure to the suction port to assist the ingress of hydraulic fluid on demand.

In one implementation of this aspect of the invention, the drive line forms part of the drive train of a vehicle. Most preferably, the drive shaft of the motor/pump unit is effectively integral with the drive line of the vehicle. In this case, the connection means may simply include a pair of universal joints allowing one sub-section of the drive line to be constituted by the drive shaft of the pump/motor unit. In other embodiments, however, it will be appreciated that gears, clutches, or other connection means may be interposed to transmit rotary drive between the vehicle drive line and the pump/motor unit. Independently of the coupling mechanism which is essentially internal to the pump/motor unit, such external connection means may be mechanical, hydraulic, pneumatic or electromagnetic. They may also be permanently engaged or decouplable, manual or automatic, and may include constant or variable reduction ratios.

Preferably, the pump/motor assembly includes at least three external ports to permit ingress and egress of hydraulic fluid, with a first port communicating with an

inlet of the hydraulic reservoir, a second port communicating with an outlet of the hydraulic reservoir, and a third port communicating with the accumulation means. A heat exchanger is preferably disposed between the first port and the hydraulic fluid reservoir.

In one embodiment of the invention, a plurality of positive displacement axial piston pumps is arranged axially along a single drive shaft or drive line. These pumps may be connected hydraulically to operate in series, parallel, or a combination of both.

Preferably, the energy management system includes a flow control circuit through which hydraulic fluid may be selectively directed, the control circuit being adapted to provide a controllable resistance enabling the pump/motor unit selectively to exert a retarding force on the drive shaft when required, even if the accumulation means are fully charged.

Preferably, the accumulation means include a gas/liquid accumulator comprising a double-ended cylinder and a piston adapted to float sealingly within the cylinder. One side of the cylinder preferably contains a compressible inert gas such as nitrogen, while the other side of the cylinder is preferably connected hydraulically to the pump/motor unit. The accumulator is preferably thereby adapted to store energy by pumping hydraulic fluid into one side of the cylinder, so as to compress the gas on the other side by displacement of the floating piston, and subsequently to release that energy by expulsion of hydraulic fluid as the compressed gas expands. In alternative embodiments, however, other forms of accumulator, such as bladder, bellows or diaphragm type accumulators, may be readily substituted. The assembly optionally includes a plurality of accumulators, which may be selectively connected in series, parallel or a combination of both, as required.

BRIEF DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the invention will now be described, by way of example only, with reference to the accompanying drawings in which:-

figure 1 is a cross-sectional view showing a pump/motor assembly incorporating a coupling mechanism including an epicyclic gear train, according to a first embodiment of the invention;

figure 2 is an enlarged cross-sectional view showing in more detail the decoupling mechanism incorporating the epicyclic gear train from the pump/motor unit of figure 1;

figure 3 shows an alternative decoupling mechanism, incorporating a different configuration of epicyclic gear train, according to a second embodiment of the invention;

figure 4 shows an alternative decoupling mechanism according to a third embodiment of the invention;

figure 5 shows an alternative decoupling mechanism, according to a fourth embodiment of the invention;

figure 6 shows an alternative decoupling mechanism, according to a fifth embodiment of the invention;

figure 7 shows an alternative decoupling mechanism, according to a sixth embodiment of the invention;

figure 8 shows an alternative decoupling mechanism, according to a seventh embodiment of the invention; and

figure 9 is a diagrammatic perspective view showing a pump/motor assembly of the general type illustrated in figures 1 to 8, incorporated into a road vehicle as part of a regenerative drive system (RDS), according to a further aspect of the invention.

PREFERRED EMBODIMENTS OF THE INVENTION

Referring initially to figure 1, the invention provides a positive displacement hydraulic pump/motor unit 1. The pump/motor unit includes a stationary housing 2 and a cylinder block 3 supported within the housing for rotation about a central axis 5. The block 3 incorporates a circular array of hydraulic cylinders 6 uniformly disposed in parallel relationship about the central rotational axis 5. A corresponding array of axial pistons 10 is reciprocably disposed within the respective cylinders.

A central drive shaft 12 extends through a complementary bore 13 formed in the cylinder block. The shaft 12 is drivingly connected to the cylinder block 3 by coupling means including a splined drive sleeve 14 to effect rotation of the block about the central axis, as described in more detail below.

A stationary drive plate in the form of swash plate 15 is disposed at one end of the cylinder block (the left-hand end when viewing the drawing). The swash plate is pivotably supported on a cradle within the housing, for adjustable movement within a predetermined range, about an axis substantially normal to the rotational axis of the cylinder block.

A hold-down plate 16 is disposed to locate and maintain the flanged free ends of the pistons in the appropriate relative spatial relationship, while the proximal end faces of the pistons are formed with followers 18 adapted to engage and slidably traverse the operative surface of the swash plate. In this way, rotation of the cylinder block effects sequentially staggered reciprocation of the pistons, with the amplitude of piston travel being determined by the selected angle of inclination of the swash plate.

A stationary valve plate 20 is disposed at the opposite end of the cylinder block (the right-hand end when viewing the drawings) and is rigidly bolted to the housing. The valve plate includes a valve face 21 adapted for sliding rotational engagement with a complementary mating valve face 22 formed on the abutting end of the cylinder block. The valve plate includes inlet ports (not shown) adapted for communication with a source of hydraulic fluid, and outlet ports (also not shown) adapted for fluid communication with an hydraulic load. The cylinder block is biased toward the valve plate by a hold-down spring 28, which applies a substantially constant preload force to minimise leakage across the sliding valve faces.

The valving is arranged such that hydraulic fluid is progressively drawn into the cylinders in sequence through the inlet ports as the pistons withdraw away from the valve plate and is subsequently expelled from the cylinders through the outlet ports as the respective pistons are progressively advanced toward the valve plate, under the influence of the swash plate.

The swash plate is pivotably supported within the housing such that the effective angle of inclination with respect to the rotational axis of the cylinder block is adjustable to provide selectively variable flow characteristics. In particular, the swash plate may be alternately inclined in a positive and a negative sense, thereby enabling the assembly selectively to operate as a motor or a pump. In this regard, it should be appreciated that the particular valve ports which function as inlets to the cylinders of the pump/motor unit, and those which function as outlets, will alternate according to

the operational mode of the unit. Importantly, the swash plate can also be orientated in an intermediate or neutral position in a plane effectively normal to the central axis, such that rotation of the cylinder block produces no reciprocation of the pistons. In this mode, the pump/motor unit induces no net fluid flow into or out of the cylinders, and consequently transfers no significant hydraulic load to or from the shaft.

The essential elements of construction and the basic principles of operation are common to most positive displacement axial piston hydraulic pumps, and being well understood by those skilled in the art, need not be described in more detail.

As best seen in figure 2, the pump/motor unit further includes coupling means in the form of an epicyclic gear train 30, effectively interposed between the drive shaft and the cylinder block. The gear train includes a sun gear 32 and spline formations 33 machined into the outer surface of, or otherwise fixedly connected to, the drive shaft 12 in axially spaced relationship. A plurality of planetary gears or pinions 34 is disposed around the drive shaft, rotatably supported on respective pinion shafts 35 by carrier 36, and in meshing engagement with the sun gear 32. A peripheral ring gear 38 is disposed around the planetary gears 34, in meshing engagement with them. The ring gear 38 includes an inner sleeve 40, which is slidably disposed in coaxial relationship around a corresponding first inner sleeve 42 formed on the carrier, to allow relative rotation between the ring gear and the carrier. The first carrier sleeve 42 is keyed or splined to the outer surface of one end of the drive sleeve 14, which surrounds the main drive shaft 12. The opposite end of the drive sleeve 44 is keyed or splined to the rotary cylinder block (see figure 1).

A disk clutch assembly 50 incorporates a clutch housing 52 including a first inner sleeve 54 in meshing engagement with the adjacent spline formations 33 on the input shaft. The clutch assembly also includes a second inner sleeve 56 formed on the carrier. The clutch housing 52 supports outer clutch plates 57, while the second carrier sleeve 56 supports inner clutch plates 58 such that the clutch is effectively interposed between the drive shaft and the carrier. The respective sets of radially overlapping, mutually interleaved clutch plates are selectively urged into fictional engagement by an annular hydraulic piston and cylinder assembly 59 whereby upon engagement of the clutch, drive is transmitted directly from the input shaft to the carrier, but upon disengagement, no such drive is transmitted.

A first disk brake assembly 60 incorporates a plurality of annular brake disks 62 extending radially inwardly from the stationary housing of the pump/motor unit, and a corresponding plurality of brake pads 64 extending radially outwardly from the outer surface 66 of the ring gear, such that the brake is effectively interposed between the ring gear 38 and the housing. The respective sets of radially overlapping, mutually interleaved brake disks and pads are selectively urged into fictional engagement by an annular hydraulic piston and cylinder assembly 68 whereby upon engagement of the brake, the ring gear is effectively locked to the housing but upon this engagement, the ring gear can rotate freely within the housing.

The hydraulic clutch and brake pistons are selectively regulated using suitable hydraulic control circuitry, which forms part of the overall system controller. Intermediate or hybrid states, and smooth transitions between the primary states, can be achieved by partial and progressive engagement and disengagement of the brake and clutch respectively.

By selective actuation of the clutch and brake in different combinations, it will be appreciated that different coupling and decoupling modes can be achieved. For ease of explanation, in the following exemplification the main drive shaft 12 will be referred to as the input shaft and the drive sleeve 14 will be referred to as the output shaft (noting that these elements are represented more diagrammatically in figure 2). It should be understood, however, that in practice, these input and output functions will alternate between the respective drive elements according to whether the unit is operating as a motor or a pump.

Referring then to figure 2, when the brake 60 is disengaged, the ring gear 38 is able to rotate relative to the housing, and with the clutch engaged, the carrier is effectively locked to the input shaft. In this mode, drive is transmitted from the input shaft to the carrier through the second carrier sleeve 56, and from the carrier to the drive sleeve 14 through the first carrier sleeve 42, in a 1:1 ratio. This is the direct forward drive mode. It is also the most fundamental of the coupled modes, replicating in essence the transmission mode that would prevail if the cylinder block were rigidly attached to the drive shaft, as in the prior art.

If the brake 60 is applied and the clutch disengaged, the ring gear is effectively anchored to the housing, while drive is transmitted from the sun gear 32 on the input

shaft, through the pinions, to the carrier. The carrier in turn transmits this drive to the output shaft 14 through the first carrier sleeve 42, which is splined to the output shaft. In this mode, the output shaft will rotate in the same direction as the input shaft, but at a reduced output speed. The extent of this reduction will correspond to the ratios of teeth on the respective gears, which are selected to suit the particular application. This is the forward speed reduction drive mode.

If both the brake and the clutch are disengaged, no drive is transmitted from the input shaft through the clutch to the carrier, and the ring gear 38 is free to rotate around the first carrier sleeve 42, such that no drive is transmitted from the input shaft to the output shaft. This is the decoupled mode of the coupling mechanism.

It will be appreciated that this decoupled mode, when activated in situations where the pump/motor assembly is not under load, effectively eliminates frictional drag between the cylinder block and the valve plate. It also eliminates drag between the cylinder block and other stationary elements within the housing. This feature is particularly advantageous in applications where the cylinder block rotates within an oil bath and consequently, the hydrodynamic drag on the rotational group is relatively high. For these reasons, the decoupled mode complements the neutral or "freewheeling" operational mode of the motor/pump unit, as described below.

A second embodiment of the invention is shown in figure 3, wherein similar features are denoted by corresponding reference numerals. In this example, the clutch assembly of the first embodiment is omitted, and the disk brake 60 operates on an outer sleeve 70 of the carrier 36, while the inner sleeve 40 of the ring gear 38 is keyed or splined directly to the proximal end of the output shaft or drive sleeve 14.

By this means, it will be appreciated that with the disk brake engaged, the carrier will be locked to the housing and so will not be free to rotate. In this mode, drive from the input shaft 12 is transmitted through the sun gear 32 to the planetary pinions 34 and thence from the pinions, through the ring gear to the output shaft. This will produce a reduction in speed, but in the reverse direction, and is therefore referred to as the reverse speed reduction drive mode. With the brake disengaged, the carrier is free to rotate and on this basis, no drive will be transmitted. Accordingly, this is another implementation of a decoupled mode.

~~A third of embodiment of the epicyclic coupling mechanism is~~ shown in figure 4, where again like features are denoted by corresponding reference numerals. In this example, the first disk brake assembly 60 operates on the outer surface 66 of the ring gear, as in the first embodiment, while the inner sleeve 40 of the ring gear is keyed or splined directly to the output shaft, as in the second embodiment. Further, a second disk brake assembly 72 operates on the outer sleeve 70 of the carrier, under the influence of an associated double-acting hydraulic piston and cylinder assembly 73.

By this means, with the first brake 60 engaged and the second brake 72 disengaged, no drive is transmitted and additionally, the output shaft is positively locked by the ring gear. This is the locked mode. Conversely, with the first brake 60 disengaged and the second brake 72 engaged, the carrier is locked to the housing, but the ring gear is free to rotate. In this configuration, with drive transmitted through the sun gear, pinions and ring gear, the output shaft or drive sleeve 14 will rotate in the reverse direction, at reduced speed. Accordingly, this is the reverse speed reduction mode.

Figure 5 shows a fourth embodiment of the coupling mechanism, where once again corresponding features are denoted by corresponding reference numerals. This implementation is somewhat similar to that illustrated in figure 3. In this case, however, the disk brake is replaced by a more compact band brake assembly 74, including a series of selectively engageable friction bands 76 extending circumferentially around the outer sleeve 70 of the carrier 36. With the brake bands engaged, the carrier is locked to the housing, and drive is transmitted through the sun gear, planetary gears and ring gear to the output shaft or drive sleeve 14, with consequential speed reduction and reversal of direction. With the band brake disengaged and the ring gear able freely to rotate, the output shaft is effectively decoupled from the input shaft.

Figure 6 shows a fifth embodiment of the invention, in which the disk brake 60 operates on the outer surface or sleeve 66 of the ring gear 38, the inner sleeve 42 of the carrier is keyed to the output shaft, and the carrier is supported in sliding rotational engagement with the ring gear. In this configuration, with the disk brake engaged, the ring gear is locked relative to the housing, and drive from the input shaft is transmitted through the sun gear, pinions and carrier to the output shaft in the same direction, but

at reduced speed. This is another forward speed reduction drive mode. Release of the brake again produces a decoupled mode.

Figure 7 shows a sixth embodiment of the coupling mechanism, which in essence operates in the same manner as that illustrated in figure 6, but with braking of the ring gear achieved by means of a band brake assembly 74 in place of a disk brake, in order selectively to effect the forward drive speed reduction and decoupled modes, as previously described.

Figure 8 shows a seventh embodiment of the system, which operates in a manner analogous to that of the first embodiment as described, except in that a band brake 74 is provided in place of the previous disk brake assembly.

It should be appreciated that the foregoing examples of epicyclic gear coupling mechanisms are only some of the available possibilities. It is envisaged that other configurations operating in various modes, could also be used, with different types and combinations of clutches, brakes, viscous couplings, torque converters, fluid drives and other transmission elements selectively to produce the transmission characteristics and decoupling functionality as required according to the particular application.

The present invention is particularly well adapted for implementation as part of a regenerative drive system (RDS) of the type previously described by the present applicant in the earlier application No. PCT/AU99/00740. An example of one such implementation in a vehicle chassis 80 is illustrated in figure 9.

Referring to figure 9, the hydraulic pump/motor assembly 1, as described above, forms part of an energy management system 82, wherein the drive shaft 12 of the pump/motor unit is connected to the drive line or power train 84 of the vehicle via yokes 86 and 88. In situ, these yokes form parts of universal joints at the respective ends of the drive shaft. In this way, the drive shaft of the pump/motor unit becomes serially connected with, and an integral part of, the drive line of the vehicle, thereby obviating the need for interconnecting gearboxes, chain drives, belt drives, or other intermediate transmission mechanisms. This makes the unit efficient, compact, reliable, and readily retrofitable to existing vehicles.

In this implementation, the system further includes energy accumulation means in the form of a pair of gas/liquid accumulators 90, each comprising a double-

ended cylinder 92 and a piston (not shown) adapted sealingly to float within the cylinder. One side of each cylinder contains a compressible inert gas such as nitrogen, while the other side of the cylinder is in fluid communication with the pump/motor unit via hydraulic lines. Each accumulator is thereby adapted to store energy by receiving pressurised hydraulic fluid into one side of the cylinder so as to compress the gas on the other side, and adapted subsequently to release that energy by expulsion of the hydraulic fluid as the compressed gas is allowed to expand. This method of energy accumulation and regeneration is well understood by those skilled in the art, and is described in more detail in PCT/AU99/00740. Again, however, it should be emphasised that alternative forms of energy accumulator such as bladder, bellows or diaphragm type accumulators can readily be substituted and further, that any suitable number of accumulators may be used.

In use, the system is selectively operable in any one of three primary modes. In a first braking mode, the system operates to retard the drive shaft of the vehicle by pumping hydraulic fluid into the accumulators and thereby compressing the contained gas medium. Alternately, the system is operable in a driving mode to supply supplementary power to the drive shaft of the vehicle using the pressurised hydraulic fluid from the accumulators. In the braking mode, it will be appreciated that the hydraulic pump/motor unit operates as a pump powered by the vehicle drive shaft, whereas in the driving mode, the unit operates as a motor powered by pressurised hydraulic fluid from the accumulators. The system is also operable in a third neutral or "free wheeling" mode, whereby the drive line of the vehicle is substantially unaffected by the pump/motor unit, aside from any residual frictional drag which the present invention in its preferred form aims to minimise.

In the neutral or free wheeling mode, the complementary decoupled mode of the epicyclic gear train may be used to particular advantage, by internally disengaging the drive shaft from the motor/pump unit. As previously described, this avoids the need for the cylinder block to rotate in unison with the drive shaft (and the vehicle drive line) when not required. The effect of this disengagement is to substantially eliminate a primary source of hydrodynamic drag, which arises by virtue of the fact that the rotational group normally spins in an oil bath contained within the housing. Advantageously, the decoupled mode of operation also avoids relative rotation

between the cylinder block and the valve plate face, which is another significant source of frictional loss and operational inefficiency, and which serves no purpose in the neutral mode when the pump/motor unit is doing no work. A secondary benefit of this is that because the mating valve faces do not rotate relative to one another in the decoupled mode, there is no need for continuous hydrodynamic lubrication which would otherwise be required even in the neutral mode. This in turn obviates the need for a residual level of hydraulic pressure, which would otherwise need to be continuously generated in the neutral mode by a supplementary charge-pump, a non-zero swash plate angle, or the accumulators themselves, all of which further reduce operational efficiency. A further follow-on effect is that wear at the valve faces, where fine tolerances are important in order to avoid leakage flow, is minimised. The decoupled mode may also be used to prevent rotation of the cylinder block above a predetermined threshold speed, so as to avoid excessive noise, vibration, internal stress or other critical design parameters.

The three primary operational modes are controlled by the angle of inclination of the swash plate 15, in the manner previously described. This angle, together with the related operation of the various clutches and brakes associated with the epicyclic gear train, is regulated by a computer controlled electronic RDS management system 100. The RDS management system is programmably responsive to a predetermined series of system parameters such as accelerator, brake and clutch pedal positions, engine speed, gear selection, engine manifold pressure, swash plate position, drive line torque, accumulator pressure and hydraulic pump/motor pressure. The system may also be pre-programmed with topographical mapping and terrain data, thereby enabling it effectively to anticipate inclines and declines as well as stopping and acceleration points on known routes, and optimise the performance of the RDS on that basis. Again, the general operating principles in this respect are described in more detail in PCT/AU99/00740.

The RDS unit as illustrated is positioned in the central drive line of the vehicle, immediately downstream of the engine 102 and gearbox 104. This is advantageous, because in that position, the system can be readily retrofitted to existing vehicles by replacement of a standard section of the original drive line. It should be understood, however, that the unit may alternatively be positioned between the engine and

gearbox, for example in direct connection with the crank shaft. As another alternative, it may be incorporated into a gearbox or engine casing. It may also be positioned downstream of one or both of the differentials 108, and may even be incorporated into an axle. In that case, it will be apparent that several RDS units may be incorporated into multiple drive line sections, axles or stub axles.

The epicyclic coupling mechanism of the present invention provides substantial additional functionality in terms of the manner in which drive is transmitted from the drive line or drive shaft to the pump. In this regard, relative rotational speed as between the main drive shaft and the cylinder block can be increased or decreased and the direction reversed as required to sit particular applications. These different drive modes can be pre-set or selectively engageable, depending upon the level of functionality and sophistication required of the system. Furthermore, smooth transitions between different modes can be reliably achieved by progressive modulation of the clutches and brakes. while retaining the inherent benefits of direct drive transmission and coaxial integration between the drive line and pump/motor unit.

Furthermore, using the additional functionality of provided by the decoupling mode, the present invention provides an efficient and effective mechanism for substantially reducing frictional or hydrodynamic drag under off-load conditions in axial piston hydraulic pumps and motors, of almost any configuration and in virtually any application. Importantly, these improvements can be achieved without compromising sealing performance under heavy load or high pressure conditions. The benefits of this are particularly substantial in the context of regenerative drive systems, wherein the pump/motor unit may be required to operate in a neutral or freewheeling mode for prolonged periods, and any residual frictional drag during such periods can impact cumulatively on the overall efficiency of the system. In all these respects, the invention represents both a practical and a commercially significant improvement over the prior art.

Although the invention has been described with reference to specific examples, it will be appreciated by those skilled in the art that the invention may be embodied in many other forms. For example, while the particular implementation of the invention shown in the drawings is described predominantly in the context of

heavy road going vehicles, it will also be appreciated that regenerative drive systems of this type can be readily adapted to practically any environment in which it is advantageous to store excess mechanical energy at some time, for use at a later time. Typical examples of other potential applications include elevators and lifts, escalators and travelators, conveyors, cranes and other hoisting devices, as well as other forms of transportation such as rail, shipping and aeronautical applications. Furthermore, the application of the invention is not limited to regenerative drive systems, but is more broadly applicable to virtually any application in which hydraulic motors and pumps are used. Also, the invention is not limited to positive displacement or axial piston hydraulic pumps, but could also, for example, be adapted for use with vane, gear and radial piston pumps.

CLAIMS

1. A positive displacement hydraulic pump/motor assembly including:-

a rotary cylinder block having a central axis and incorporating a generally circular array of cylinders disposed around the axis;

a corresponding plurality of pistons reciprocably disposed within the respective cylinders;

drive means including a drive shaft extending through a bore formed in the cylinder block to effect rotation of the cylinder block about the central axis;

a drive plate disposed at one end of the cylinder block to effect sequentially staggered reciprocation of the pistons in response to rotation of the cylinder block;

a stationary valve plate disposed at an opposite end of the cylinder block, the valve plate having a valve face adapted for sliding rotational engagement with a complementary mating face formed on the cylinder block;

the valve plate further including at least one inlet port adapted for fluid communication with a source of hydraulic fluid and at least one outlet port adapted for fluid communication with an hydraulic load;

the ports being disposed such that in use, hydraulic fluid is progressively drawn into the cylinders in sequence as the respective pistons are displaced away from the valve plate and subsequently expelled from the cylinders as the pistons are progressively displaced toward the valve plate;

the pump/motor assembly further including a coupling mechanism incorporating an epicyclic gear train operable in at least one engaged mode to connect the drive shaft to the cylinder block.

2. A pump/motor assembly according to claim 1, wherein the cylinders are disposed in generally parallel relationship with respect to the central axis.

3. A pump/motor assembly according to claim 1 or claim 2, wherein the drive plate takes the form of a stationary swash plate, which is inclined with respect to the central axis of the cylinder block.

4. A pump/motor assembly according to claim 3, wherein floating ends of the pistons remote from the valve plate include followers adapted to traverse the swash plate as the cylinder block rotates.

5. A pump/motor assembly according to claim 4, further including a hold-down plate disposed to capture the floating ends of the pistons and retain the followers in sliding contact with the swash plate.

6. A pump/motor assembly according to any one of claims 3 to 5, wherein the angle of inclination of the swash plate is selectively adjustable, to provide variable flow rate characteristics.

7. A pump/motor assembly according to claim 6, wherein the swash plate is able to be selectively inclined in a positive or a negative sense, thereby enabling the assembly alternately to operate as a motor or a pump.

8. A pump/motor assembly according to claim 6 or claim 7, wherein the variable swash plate can be oriented in an intermediate or neutral position, effectively normal to the central axis, such that rotation of the cylinder block causes no movement of the pistons, and hence induces no substantial load.

9. A pump/motor assembly according to any one of the preceding claims, further including bias means disposed to apply a bias force urging the respective mating faces on the cylinder block and the valve plate into sealing engagement.

10. A pump/motor assembly according to claim 9, wherein the bias force applied by the bias means is selectively variable.

11. A pump/motor assembly according to any one of the preceding claims, wherein the coupling mechanism is selectively operable in at least two distinct modes, at least one of which is an engaged mode whereby drive is transmitted from the drive shaft to the cylinder block.

12. A pump/motor assembly according to any one of the preceding claims, wherein the coupling mechanism is selectively operable in a disengaged mode, to allow the drive shaft to rotate substantially independently of the cylinder block, and thereby to operate as a decoupling mechanism.

13. A pump/motor assembly according to any one of the preceding claims, wherein the epicyclic gear train includes an input shaft connected to a sun gear, a plurality of planetary gears disposed in meshing engagement around the sun gear, a planetary gear carrier, a peripheral ring gear in meshing engagement with the planetary gears, and an output shaft.

14. A pump/motor assembly according to claim 13, wherein one of the input and output shafts constitutes the primary drive shaft extending coaxially through the pump/motor unit, while the other of the input and output shafts constitutes a drive sleeve disposed coaxially around the primary shaft.

15. A pump/motor assembly according to any one of the preceding claims, wherein the coupling mechanism further includes a clutch and/or a brake assembly, being independently actuable on selected elements of the epicyclic gear train so as to transmit rotary drive between the drive shaft and the cylinder block in a plurality of different modes.

16. A pump/motor assembly according to claim 15, wherein said different modes are selected from a group comprising: a direct forward drive mode; a forward speed reduction drive mode; a forward speed increasing drive mode, a decoupled mode; a direct reverse drive mode, a reverse speed reduction drive mode; a reverse speed increasing drive mode; and a locked mode.

17. An energy management system operable in a driving mode, a braking mode and a neutral mode, the energy management system including:-

energy accumulation means operable as the hydraulic load to selectively store and release energy through controlled receipt and release of hydraulic fluid;

a positive displacement hydraulic pump/motor assembly as defined in any one of the preceding claims, in fluid communication with the energy accumulation means;

a low pressure hydraulic reservoir in fluid communication with the pump/motor assembly; and

connection means for connecting the pump/motor assembly to a drive line;

the system being arranged such that in the braking mode the pump/motor assembly retards the drive line by pumping hydraulic fluid into the accumulation means, in the driving mode the pump/motor assembly supplies supplementary power to the drive line using pressurised hydraulic fluid from the accumulation means, and in the neutral mode the pump/motor assembly is effectively inoperative and exerts no substantial driving or retarding influence on the drive line.

18. An energy management system according to claim 17, wherein the reservoir includes a low-pressure accumulator or constant pressure chamber adapted to supply a positive pressure to a suction port of the pump/motor unit.

19. An energy management system according to claim 17 or claim 18, wherein the drive line forms part of a drive train of a vehicle.

20. An energy management system according to claim 19, wherein the drive shaft of the pump/motor assembly is effectively integral with the drive line of the vehicle.

21. An energy management system according to claim 20, wherein the connection means include a pair of universal joints allowing one sub-section of the drive line to be constituted by the drive shaft of the pump/motor unit, such that the coupling mechanism provides a connection between the vehicle drive line and the cylinder block.

22. An energy management system according to any onc of claims 17 to 21, wherein the pump/motor assembly includes at least three external ports to permit ingress and egress of hydraulic fluid, with a first port communicating with an inlet of the hydraulic reservoir, a second port communicating with an outlet of the hydraulic reservoir, and a third port communicating with the accumulation means.

23. An energy management system according to claim 22, further including a heat exchanger disposed between the first port and the hydraulic fluid reservoir.

24. An energy management system according to claim 22, wherein a plurality of said pump/motor assemblies is arranged axially along a common drive shaft.

25. An energy management system according to any one of claims 17 to 24, further including a flow control circuit through which hydraulic fluid can be selectively directed, the control circuit providing a controllable resistance enabling the pump/motor assembly selectively to exert a retarding force on the drive shaft when the accumulation means are fully charged.

26. An energy management system according to according to any one of claims 17 to 25, wherein the accumulation means include a gas/liquid accumulator comprising a double-ended cylinder and a piston adapted to float sealingly within the cylinder, and wherein side of the cylinder contains a compressible inert gas and the other side of the cylinder is connected hydraulically to the pump/motor assembly.

27. An energy management system according to claim 26, wherein the accumulator is adapted to store energy by pumping hydraulic fluid into one side of the cylinder so as to compress the gas on the other side by displacement of the floating

piston, and subsequently to release that energy by expulsion of hydraulic fluid as the compressed gas expands.

28. An energy management system according to claim 27, including a plurality of said accumulators, connected in series or parallel.

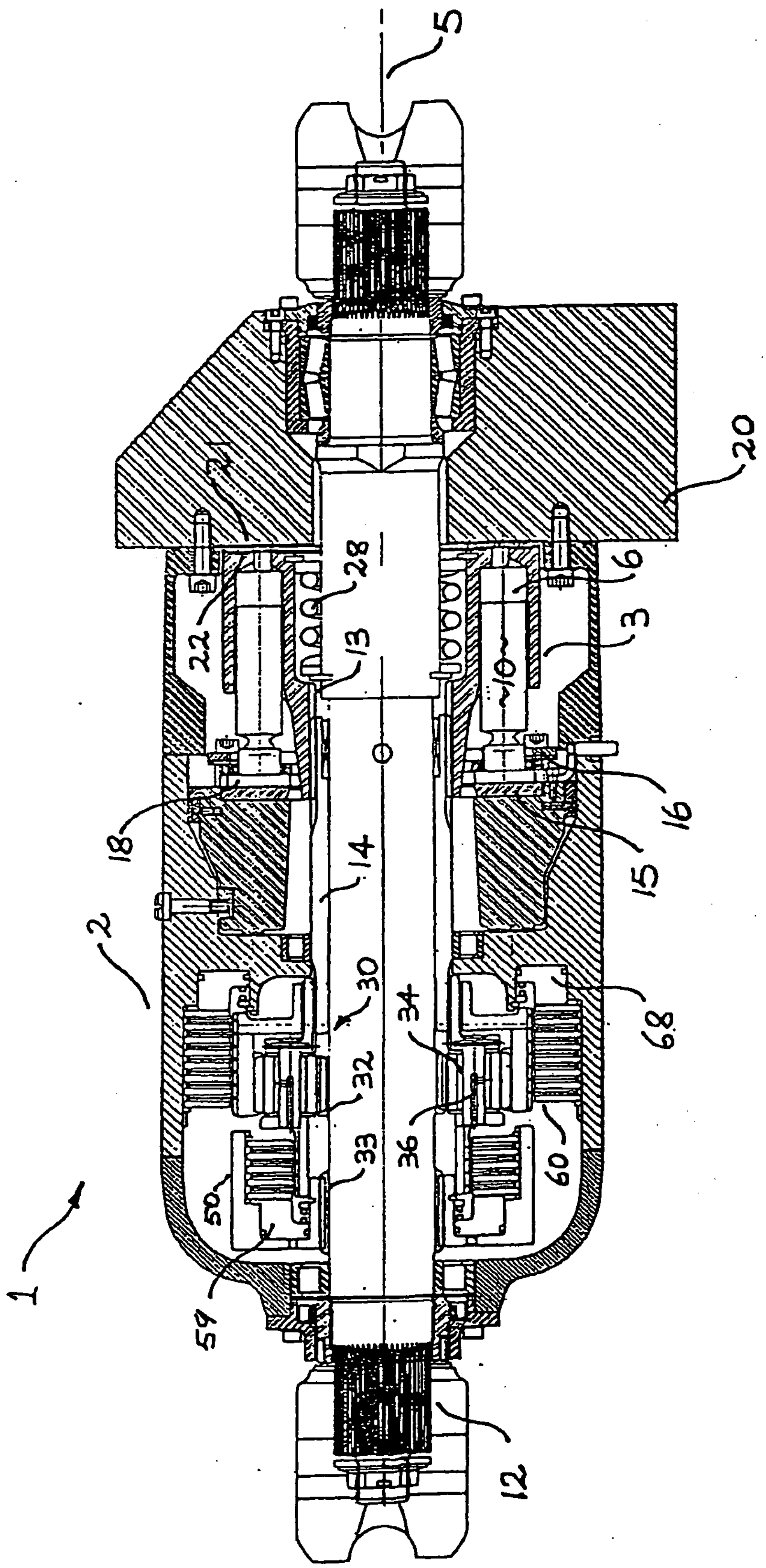


Figure 1

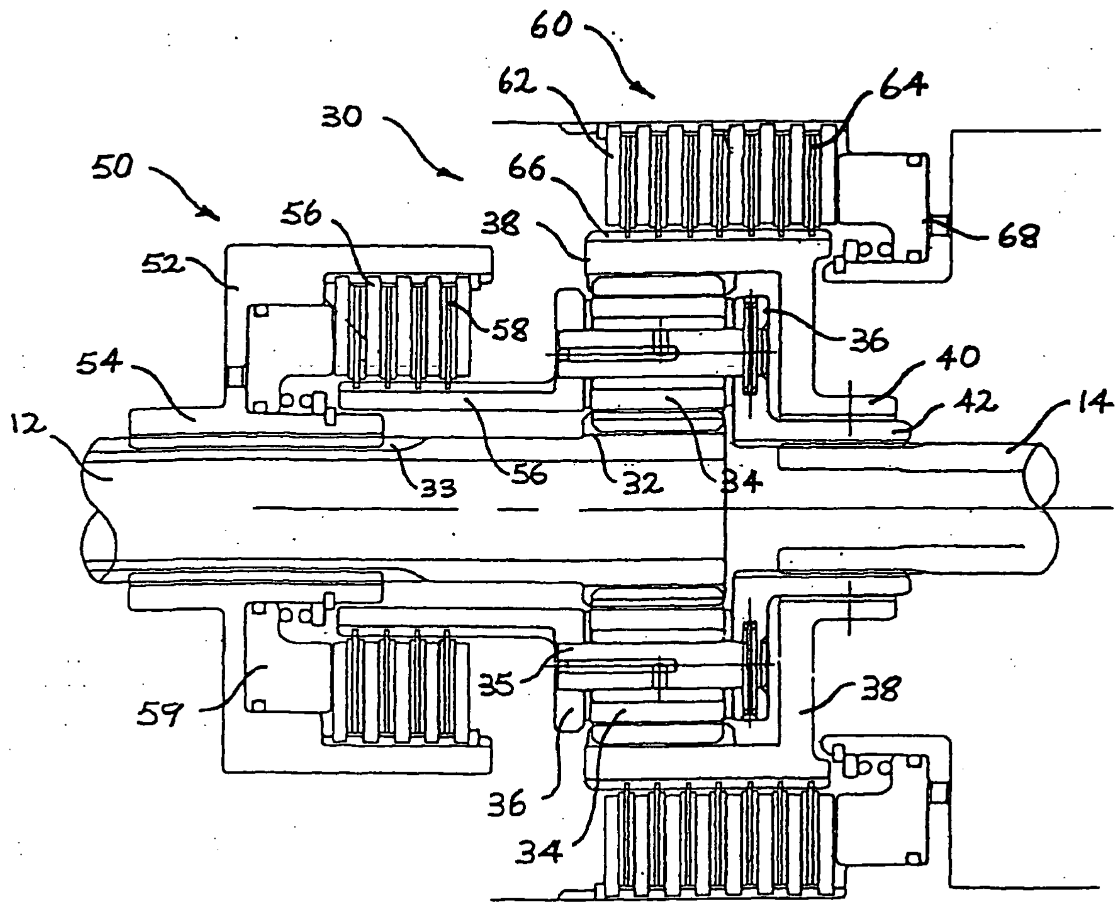

60
62
64
30
50
66
56
38
68
52
58
36
40
54
42
14
12
33
56
32
34
35
38
59
36
34

Figure 2

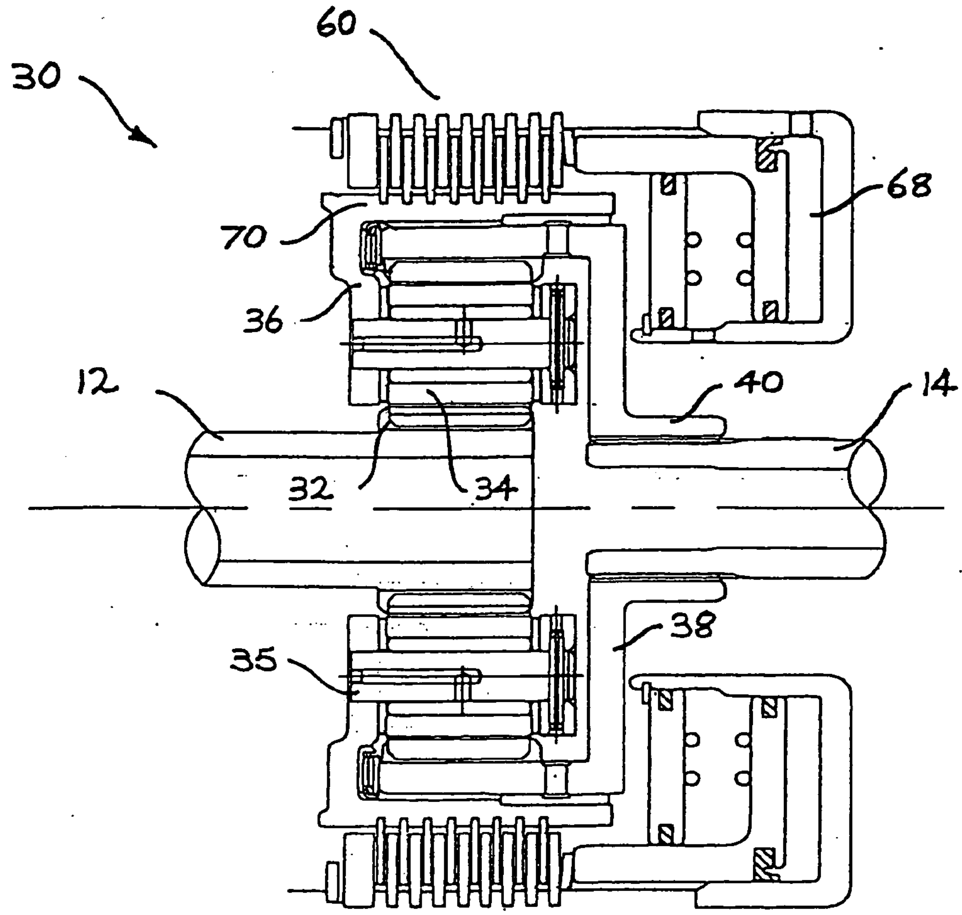


Figure 3

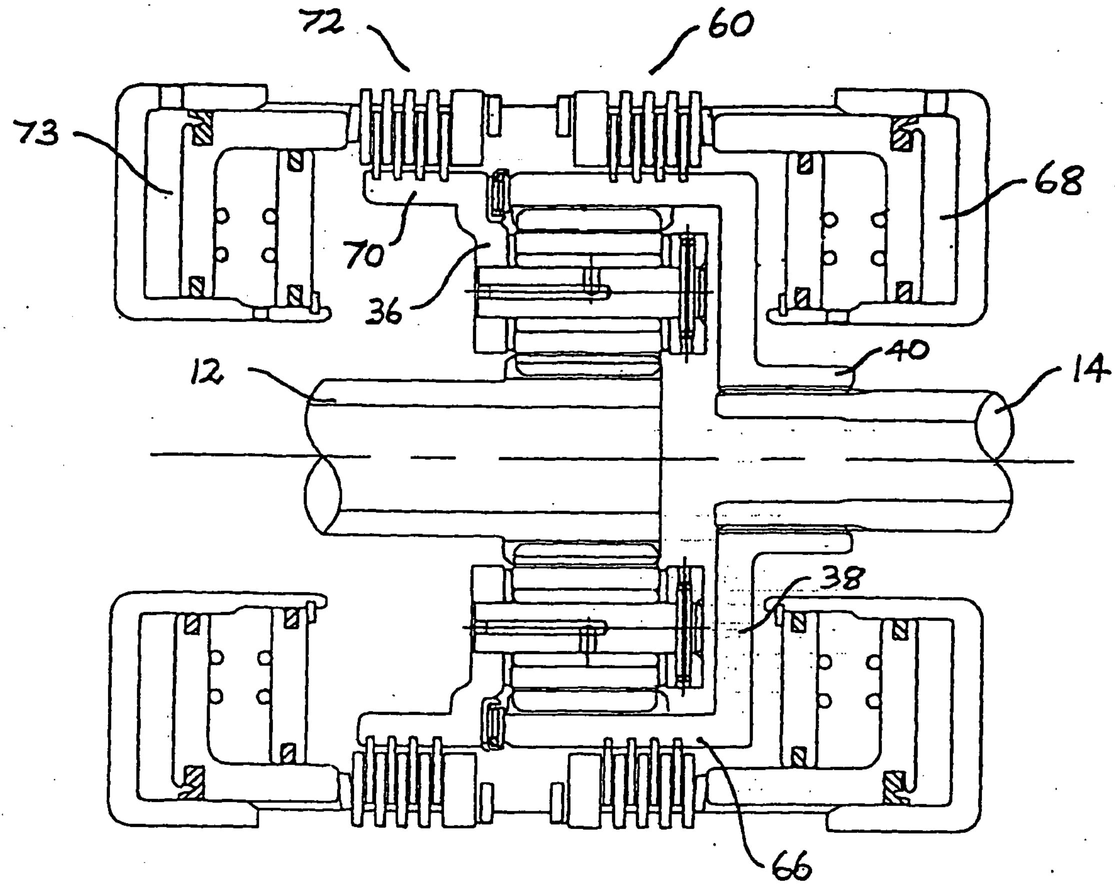


Figure 4

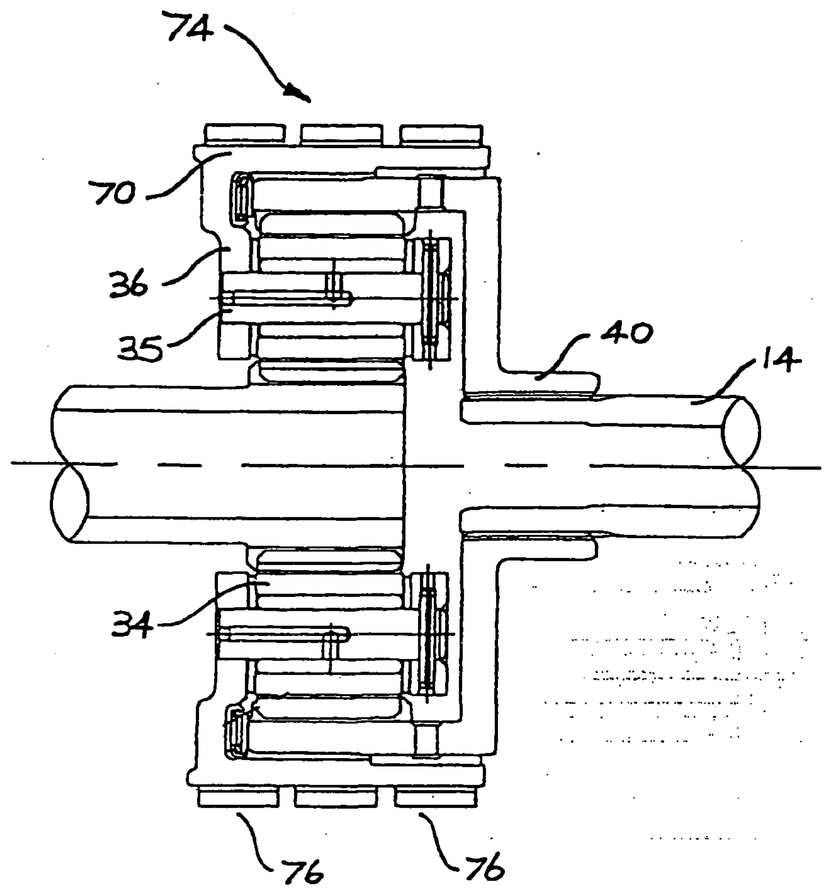


Figure 5

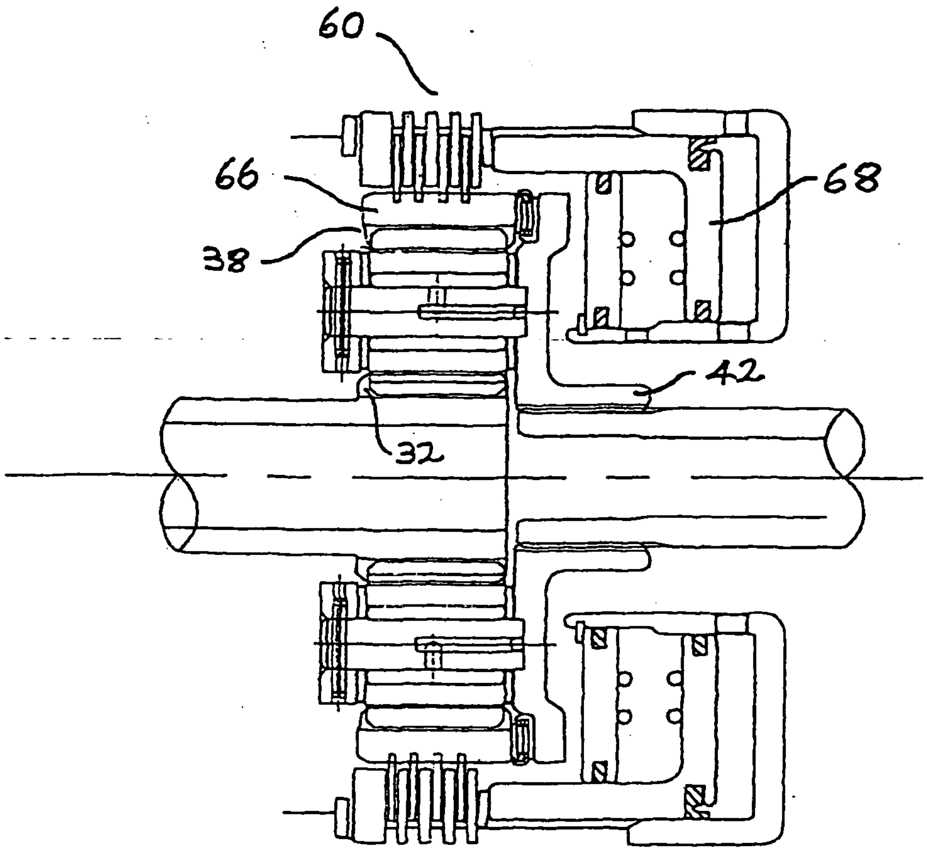


Figure 6

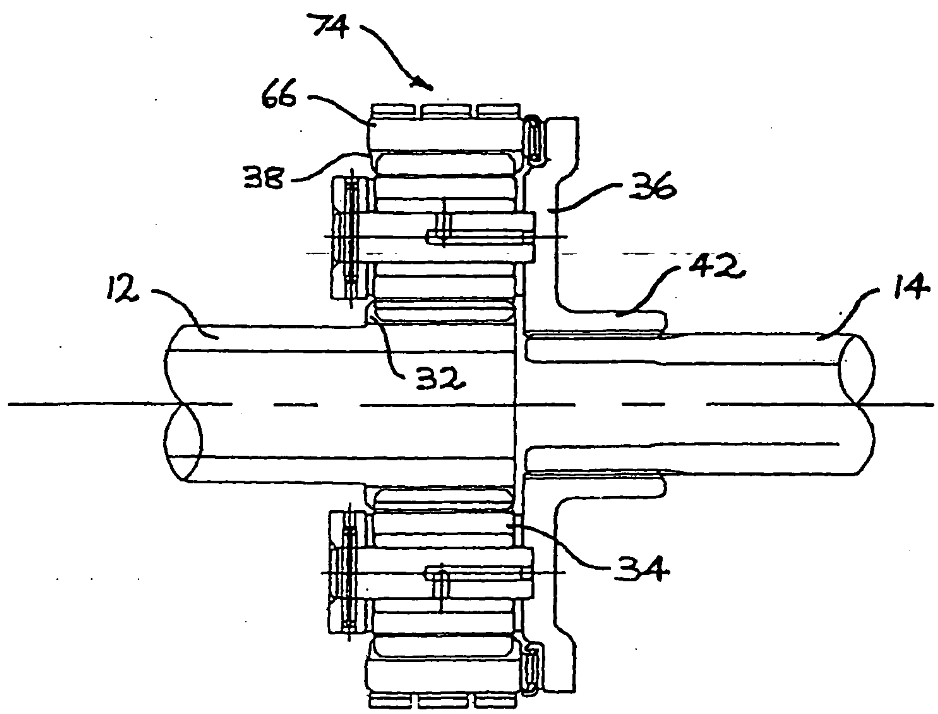


Figure 7

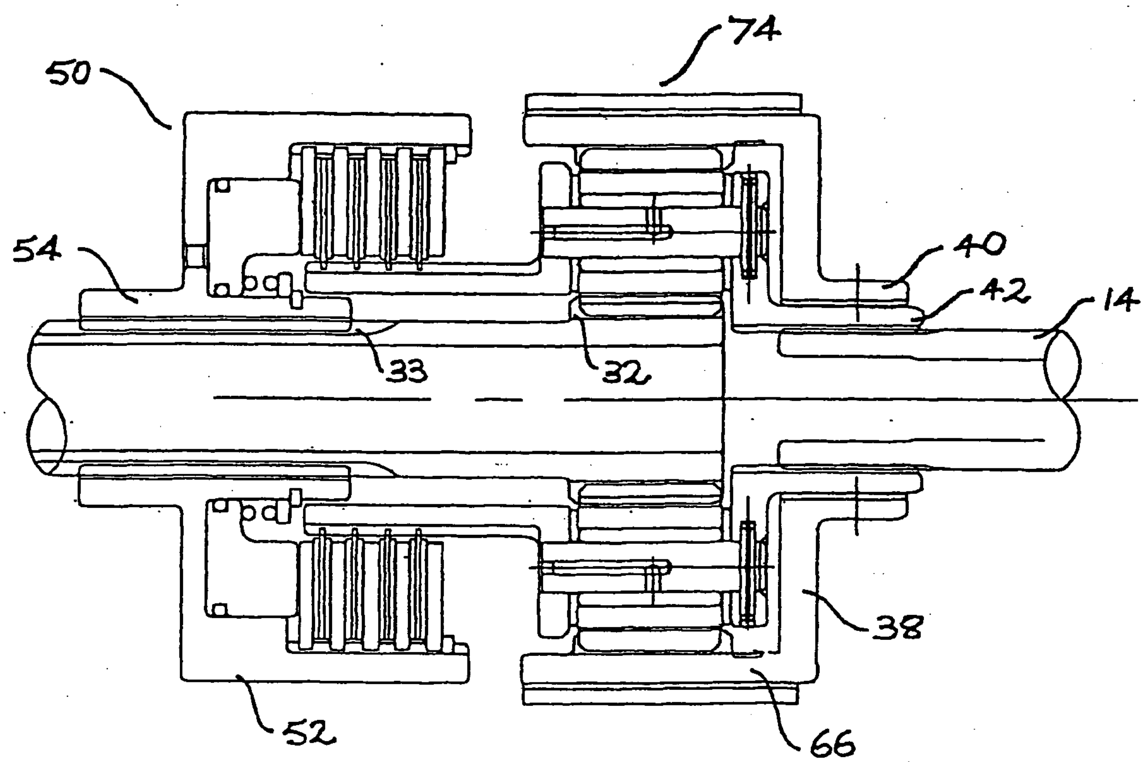


Figure 8

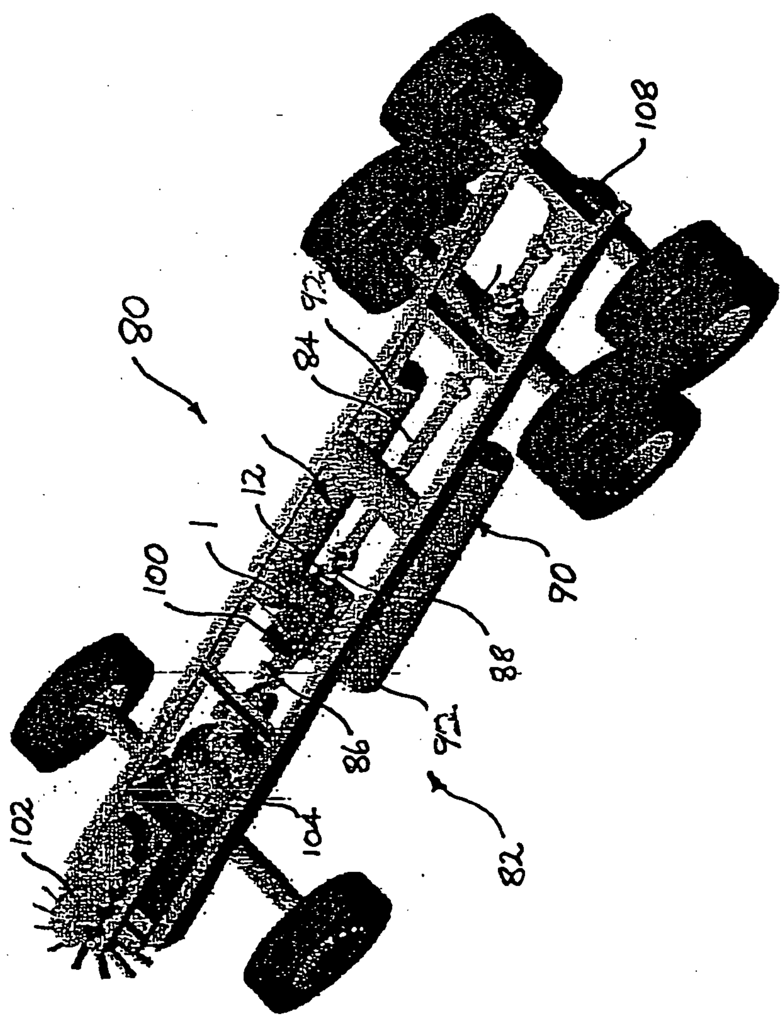


Figure 9

INTERNATIONAL SEARCH REPORT	International application No. PCT/AU03/01337

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl. [7]: F04B 1/22, 49/02; F01B 3/02, 31/22

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
See under "Electronic database consulted"

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
DWPI : with keywords: epicyclic, ring, swash, axial piston and similar terms

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	GB2264540A (LINDE AG) 1 September 1993	
X	The whole document	1-16
Y	The whole document	17-28
	DE 19910047A1 (LINDE AG) 14 September 2000	
X	The whole document	1-16
Y	The whole document	17-28
	US 5528978A (FORSTER) 25 June 1996	
X	The whole document, particularly column 3, line 1-5	1-16
Y	The whole document, particularly column 3, line 1-5	17-28

[X] Further documents are listed in the continuation of Box C [X] See patent family annex

* Special categories of cited documents:

"A" Document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent but published on or after the international filing date

"L" Document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" Document referring to an oral disclosure, use, exhibition or other means

"P" Document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
3 November 2003	11 NOV 2003

Name and mailing address of the ISA/AU	Authorized officer
AUSTRALIAN PATENT OFFICE PO BOX 200, WODEN ACT 2606, AUSTRALIA E-mail address: pct@ipaustralia.gov.au Facsimile No. (02) 6285 3929	ASANKA PERERA Telephone No : (02) 6283 2373

INTERNATIONAL SEARCH REPORT

International application No.
PCT/AU03/01337

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	GB 1535913A (EATON CORPORATION) 13 December 1978	
X	Figures 1, 2 and accompanying description	1-16
Y	Figures 1, 2 and accompanying description	17-28
	FR 2330921A1 (EATON CORPORATION) 3 June 1977	
X	Figures 1, 2 and accompanying description	1-16
Y	Figures 1, 2 and accompanying description	17-28
	Derwent Abstract Accession No. 88-291036/41, Class Q13 SU 1384421A (LIVESTOCK FODDER) 30 March 1988	
X	Abstract and Figure	1-16
Y	Abstract and Figure	17-28
	US 4986383A (EVANS) 22 January 1991	
Y	The whole document	17-28
	Note: US 4986383A above is combined with any one of the preceding documents to test inventive step of claims 17-28.	

INTERNATIONAL SEARCH REPORT Information on patent family members	International application No. **PCT/AU03/01337**

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member					
GB	2264540	DE	4206101	FR	2688852	JP	6011012
		US	5391122				
DE	19910047	JP	2000264083				
US	5528978	FR	2688041	GB	2265870	JP	6011013
GB	1535913	DE	2548473	DK	484675	ES	442337
		GB	1535911	GB	1535912	JP	51066967
		JP	53090556	JP	54059558	NO	753606
		SE	7512035	US	4040312		
FR	2330921						
US	4986383						

END OF ANNEX